ENERGY RECOVERY IN RECIRCULATION AIR SYSTEMS AND ITS CALCULATION OF DUST CONCENTRATION, RELATIVE HUMIDITY & HEATER CAPACITY

Song Ziyan [1], Tan Kunxiong [2], Chu Jiaming [1]

[1] TongJi University, Shanghai, China
[2] Hongkong Polytechnic University, Kowloon, Hongkong

ABSTRACT

In some ventilation and dust removing system of workshop, a great deal of energy is wasted due to heated and cooled indoor air being exhausted directly. So if the dust in the indoor air is removed and then recirculate, the objective of energy saving can be reached. In this paper, the calculation equation of dust concentration, relative humidity in indoor and recirculate air and heater capacity are given along with the control measures to them. This kind of system has been practiced at a plant building located in heating region of northern China, the expectable result of energy saving was obtained, thus verified the conclusion in this paper.

CONSTANT RECIRCULATE AIR VOLUME VENTILATION AND DUST REMOVING SYSTEM

This kind of system is shown in Fig.1. Most of dust emitted during the process of production is collected by capturing hood then be delivered into dust separator along with air volume V_A. The cleaned up V_P then is returned into workshop. The dust concentration C_P of V_P depends on dust emission rate ε the collection efficiency of capturing hood and penetration rate of dust separator. In addition, outdoor air volume V_N is supplied into and equal air volume V_L is exhausted from workshop, their respective dust concentration are C_N and C_L, the volume of workshop is V_R.

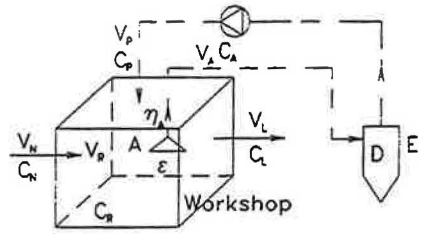

Figure 1 Constant recirculate air volume ventilation and dust removing system.

According to the mass balance equation of dust, the variation rate of indoor dust concentration C_R by time t is :

$$V_R \frac{dC_R}{dt} = \varepsilon(1-\eta_A) + V_P C_P + V_N C_N - V_L C_L - V_A C_R \qquad [\text{mg/s}]\ldots(1)$$

If outdoor air concentration is negligible, the dust contaminant is from two parts: one part is escaping rate of dust $\varepsilon(1-\eta_A)$ from capturing hood, the other is the dust rate $C_P V_P$ from recirculate air V_P. The dust concentration of recirculate air depends on the penetration rate E of dust separator.

$$V_PC_P=\varepsilon\eta_AE+V_AC_RE \qquad ...(2)$$

If it is assumed that indoor dust contaminant can be perfect diffused immediately, the exhaust dust contaminant is also from two parts: one part is the dust rate in exhaust air $V_LC_L=V_LC_R$, the other is collection rate V_AC_R by capturing hood.

Then from eq.(1), the following equation can be obtained:

$$V_R\frac{dC_R}{dt}=\varepsilon(1-\eta_A)+\varepsilon\eta_AE+V_AC_RE-V_LC_R-V_AC_R \qquad ...(3)$$

eq.(3) can be changed into:

$$\frac{dC_R}{dt}+\frac{V_A(1-E)+V_L}{V_R}C_R=\frac{\varepsilon[1-\eta_A(1\text{-}E)]}{V_R} \qquad ...(4)$$

where: $\lambda_1=\dfrac{V_A(1-E)+V_L}{V_R}$ ----theoretical air change rate [1/h] ...(5)

$$A_1=\frac{\varepsilon[1-\eta_A(1\text{-}E)]}{V_R} \qquad ...(6)$$

combining eq.(5), (6) and (4). We can get:

$$\frac{dC_R}{dt}+\lambda_1C_R=A_1 \qquad ...(7)$$

It is assumed that the emission rate of dust is constant ($d\varepsilon$ / $dt=0$) and the initial dust concentration is negligible, eq.(7) can be solved:

$$C_R=\frac{\varepsilon[1-\eta_A(1\text{-}E)]}{V_A(1\text{-}E)+V_L}(1-e^{-\lambda_1t}) \qquad ...(8)$$

In equilibrium condition, after enough long time, the exponential function $e^{-\lambda 1t}$ tends to be zero. In most of practical ventilation systems, λ_1 is 2~5[1/h], thus 2 hours later $e^{-\lambda 1t}$ is negligible, then the dust concentration can be expressed exactly by the following equation:

$$C_R=\frac{\varepsilon[1-\eta_A(1\text{-}E)]}{V_A(1\text{-}E)+V_L} \qquad ...(9)$$

VARIABLE RECIRCULATE AIR VOLUME VENTILATION AND DUST REMOVING SYSTEM

With regard to the workshop whose dust emission rate is varied greatly, the dust concentration C_P in recirculate air fluctuates along with the dust concentration C_A. in order to control the dust concentration and save energy, when C_P increases, the exhausted air volume V_L and outdoor air volume V_N need to be increased. This kind of system is shown in Fig.2.

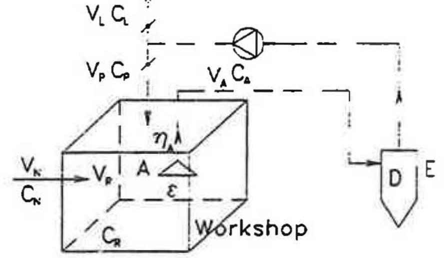

Figure 2 variable recirculate air volume ventilation and dust removing system.

Referring to Fig.2, the variation rate of indoor dust concentration by time is:

$$V_R \frac{dC_R}{dt} = \varepsilon(1-\eta_A) + V_P C_P + V_N C_N - V_A C_R \quad ...(10)$$

eq.(10) can also be changed into:

$$\frac{dC_R}{dt} + \frac{V_A(1-E)+V_L E}{V_R} C_R = \frac{\varepsilon\{1-\eta_A[1-E(1-V_L/V_A)]\}}{V_R} \quad ...(11)$$

If time t is greater than 2 hours, the approximate solution to eq.(11) is:

$$C_R = \frac{\varepsilon\{1-\eta_A[1-E(1-V_L/V_A)]\}}{V_A(1-E)+V_L E} \quad [mg/m^3]...(12)$$

From eq.(12), we can concluded that when V_A is constant, increasing the exhausted air volume V_L can lead to the decrease of the indoor dust concentration C_R and then prevent it from exceeding the dust concentration standard. If we can adjust the recirculate air volume V_P and the exhausted air volume V_L, the indoor dust concentration can be controlled effectively.

THE ALLOWABLE DUST CONCENTRATION C_P IN RECIRCULATE AIR

In variable recirculate air volume ventilation system, it is assumed that the dust concentration C_N in the outdoor air is zero. Eq.(10) can be changed into:

$$V_R \frac{dC_R}{dt} = \varepsilon(1-\eta_A) + V_P C_P - V_A C_R \quad ...(13)$$

eq.(13) can also be changed into:

$$\frac{dC_R}{dt} + \frac{V_A}{V_R} C_R = \frac{\varepsilon(1-\eta_A)+V_P C_P}{V_R} \quad ...(14)$$

If time t is greater than 2 hours, the approximate solution to eq.(14) is:

$$C_R = \frac{\varepsilon(1-\eta_A)+V_P C_P}{V_R} \quad ...(15)$$

The dust concentration in recirculate air is that in air out of dust separator, it depend on the total air volume V_A, the dust emission rate ε, the collection efficiency η_A of capturing hood and the penetration rate E of dust separator:

$$C_P = C_A E = \left(\frac{\varepsilon\eta_A}{V_A} + C_R\right)E$$

$$\varepsilon(1-\eta_A) = \frac{V_A(1-\eta_A)}{\eta_A}\left(\frac{C_P}{E} - C_R\right) \quad ...(16)$$

combing eq.(15) and (16):

$$C_R = \frac{1-\eta_A + \frac{V_P}{V_A}\eta_A E}{E} C_P \quad ...(17)$$

This expression shows the relation between the dust concentration C_P in recirculate air and the indoor dust concentration C_R. It is assumed that air leakage of the whole system is negligible and the exhausted air volume V_C is equal to 20% V_A, so the recirculate air volume V_P is equal to 80% V_A. If collection efficiency of capturing hood η_A and the penetration rate E of dust separator are supposed to be 0.98 and 0.01 respectively, from the eq.(17) we can obtain:

$$C_R=2.78C_P$$

because C_R should be lower than maximum allowable concentration C_{MAX}, then:

$$C_P<0.36C_{MAX}\cong 1/3C_{MAX}$$

Based on the above value of V_L, V_P, η_A and E, we can concluded that dust concentration C_P should be lower than 1/3 of the maximum allowable concentration C_{MAX} and 80% of the total consumed energy can be saved with perfect thermal insulation of the system.

THE RELATIVE HUMIDITY OF RECIRCULATE AIR VENTILATION SYSTEM

In the workshop with high moisture gain, the indoor air humidity ratio gradually increase due to the use of recirculate air ventilation system. So in order to maintain the indoor relative humidity within the range of acceptable standards for occupational health and production requirement. For two system mentioned above, The moisture flowchart is shown in Fig.3.

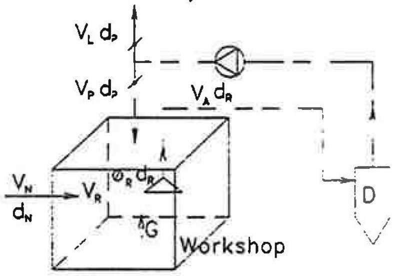

Figure 3 Moisture flowchart

Referring to Fig.3, it is supposed that the moisture gain G is well distributed and the ventilation system is perfect insulated without dew and air leakage, the humidity ratio of recirculate air is equal to that of indoor air and total air volume V_A is equal to the sum of recirculate air volume V_P and exhausted air volume V_L. If outdoor air volume V_B is equal to V_L and outdoor air humidity ratio defined as d_N, the variation rate of indoor air humidity ratio by time t is:

$$V_R\gamma_2\frac{d(d_R)}{dt}=G+V_P\gamma_2 d_P+V_N\gamma_1 d_N-V_A\gamma_2 d_R \qquad ...(18)$$

where: γ_1----outdoor air density [kg/m^3]

γ_2----indoor air density [kg/m^3]

$$d_P=d_R \qquad ...(19)$$

$$V_P=V_A-V_L=V_A-V_N \qquad ...(20)$$

The eq.(19) and (20) are put into eq.(18):

$$V_R\gamma_2\frac{d(d_R)}{dt}=G+(V_A-V_L)\gamma_2 d_P+V_N\gamma_1 d_N-V_A\gamma_2 d_R \qquad ...(21)$$

If the time t is grater than 2 hours, the approximate solution to eq.(21) can be obtained:

$$d_R=\frac{G+V_L\gamma_1 d_N}{\gamma_2 V_L} \qquad ...(22)$$

eq.(22) can be developed into:

$$V_L=\frac{G}{\gamma_2 d_R-\gamma_1 d_L} \qquad ...(23)$$

Humidity ratio expression:

$$d = 622\frac{P_b}{B-\Phi P_b} \quad ...(24)$$

where: Φ----relative humidity [%]

P_b----saturation water vapor partial pressure [Pa]

B----atmospheric pressure [Pa]

Combing eq.(24) and (23), we can get the exhausted air volume V_L expression needed to maintain indoor relative humidity:

$$V_L = \frac{G(B-\Phi_R P_{bR})(B-\Phi_N P_{bN})}{622[\Phi_R P_{bR}(B-\Phi_N P_{bN})\gamma_2 - \Phi_N P_{bN}(B-\Phi_R P_{bR})\gamma_1]} \quad ...(25)$$

where: Φ_R----relative humidity of indoor air [%]

Φ_N----relative humidity of outdoor air [%]

P_{bR}----saturation water vapor partial pressure of indoor air [Pa]

P_{bN}----saturation water vapor partial pressure of outdoor air [Pa]

If most of the air harmful & contaminants in workshop is moisture, we can adjust exhaust air value and control the relative humidity of indoor air. The function relation between humidity and exhaust air volume V_L can be expressed as:

$$\Phi_R = \frac{(G+d_N V_L \gamma_1)B}{(622 V_L \gamma_2 + G + d_N V_L \gamma_1)P_{bR}} \quad ...(26)$$

THE CALCULATION OF HEATER CAPACITY

The calculation and selection of heater capacity in this system depend on the condition of service. For the workshop of intermittent service, heating must be rapid. The calculation of heater should depend on the time during which the workshop is heated required by users. According to the conservation theory of energy:

$$\begin{aligned} C\frac{d\theta}{dt} &= \gamma q + G_E C'\theta_0 - Q' - G_E C'\theta \\ &= \gamma q + G_E C'\theta_0 - \frac{\theta-\theta'}{R}m - G_E C'\theta \end{aligned} \quad ...(27)$$

where: C----ratio of specific heats (include the heat accumulated by equipment) [kw/°C]

θ----indoor air temperature [°C]

γ----vaporization latent heat [kw/kg]

q----flow rate of water vapor into heater [kg/h]

G_E----supply air, return air volume [kg/h]

C'----specific heat of air [kw/kg·°C]

θ_0----temperature of supply air before heater [°C]

Q'----heat transfer rate through the explosure of workshop [kw/h]

θ'----outer wall temperature of explosure [°C]

R----heat resistance of explosure [m^2·h·°C/kw]

m----area of the explosure [m^2]

Eq.(27) can be developed into:

$$\frac{RC}{m+RG_E C'}\frac{d\theta}{dt} + \theta = \frac{RG_E C'}{m+RG_E C'}\left(\frac{\gamma q}{G_E C'} + \theta_0 + \frac{m}{RG_E C'}\theta'\right) \quad ...(28)$$

where we define: $T = \frac{RC}{m+RG_E C'}$ ----time constant of the control object

$$K=\frac{RG_EC'}{m+RG_EC'}$$ ----coefficient of amplification

$$\theta_y=\frac{\gamma q}{G_EC'}+\theta_0+\frac{m}{RG_EC'}\theta'$$ ----input

Then eq.(28) can be expressed as:

$$T\frac{d\theta}{dt}+\Delta\theta=K\Delta\theta_y \quad ...(29)$$

If $t=t_0=0$, $\Delta\theta_0=0$, $\Delta\theta'=0$, then $\Delta\theta_r=\gamma q/G_EC'$. When the heating system starts, the flow rate of water vapor into heater is q, whose amplitude as a transition heat rate is defined as $M=\Delta\theta_{r_0}$. The solution to eq.(29) is:

$$\Delta\theta=KM(1-e^{-t/T}) \quad ...(30)$$

As expressed by eq.(30), the relation between temperature increment $\Delta\theta$ and time t can be described as exponential curve. The calculation equation of heater capacity can be obtained by eq.(30) with $M=\gamma q/G_EC'$:

$$Q=\frac{G_EC'\Delta\theta}{K(1-e^{-t/T})} \quad [kw/h]...(31)$$

where: $\Delta\theta$----the maximum increment of indoor air temperature [°C]

The time constant and coefficient of amplification in eq.(31) can be expressed as the following equations by using that of thermostatic room.

$$T=90/N \quad [min]$$

$$K=\frac{1}{1+\frac{52}{N}(\frac{1}{a}+\frac{1}{b}+\frac{1}{h})} \quad [°C/°C]$$

where: N----air change rate

a, b, h----length, width and height of workshop [m]

Referring to eq.(31), we can conclude that shorter heating time t results in higher capacity of heater. For example, if air change rate N of workshop is 5, the constant of time is:

$$T=90/N=90/5=18 \quad [min]$$

Apparently we can see that with the heater capacity calculated by using 3 times constant of time, it will take about 54 minutes to elevate indoor temperature to that as expected.

THE ENERGY SAVING OF RECRICULATE AIR VENTILATION AND DUST CONTROL SYSTEM

In the frigid zone of northern China with high temperature difference between outdoor and indoor air, while the ordinary ventilation and dust control system are running, a large quantity of heated air is exhausted and outdoor cold air is supplied. So considerable energy is wasted. Adopting recirculate air ventilation system can contribute to energy save while guaranteeing indoor dust concentration below the acceptable standard for health. So far this kind of system has been set up in some plants. It was estimated that the average air flow rate of a ventilation system is about 10000m^3/h and more than 1000 new ventilation systems will be set up. It is also assumed that the economic effect by adopting this system is ￥ 8000 and ￥ 4700 for frigid and cold zone respectively.